July 15, 2005

DECLARATION

The undersigned, Dana Scruggs, having an office at 8902B Otis Avenue, Suite 204B, Indianapolis, Indiana 46216, hereby states that she is well acquainted with both the English and German languages and that the attached is a true translation to the best of her knowledge and ability of PCT/DE 2004/001128 (INV.: HOEFS, R.).

The undersigned further declares that the above statement is true; and further, that this statement was made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code and that such willful false statements may jeopardize the validity of the application or document or any patent resulting therefrom.

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10/542815
JC12 Rec'd PCT/PTC 20 JUL 2005

# ELECTRICAL MACHINE

## Background Information

The present invention relates to an electrical machine, in particular a generator for motor vehicles, according to the general class of the independent claim.

An electrical machine designed as an alternator is made known in DE 198 04 328 A1, in the case of which the generator shaft is supported by a movable bearing in the hub of a housing part. A spring element inserted in the hub exerts a load on the outer ring of the movable bearing with an axial force to achieve a defined rolling motion of the rolling elements in the movable bearing and thereby attain a longer service life of the movable bearing. The design of this bearing arrangement is relatively complex, since a number of components is required to exert an axial load on the roller bearing. In addition, every single generator and its bearing point must be adjusted to a defined initial axial load using a calibration step. Moreover, with a bearing arrangement of this type, only a relatively steep increase in force in the axial direction can be achieved. Due to longitudinal vibrations of the rotor, then, a relatively steep increase in spring force results and the load on the bearing therefore increases steeply. A steep increase in the spring load and the resultant high overall axial force load on the bearing is disadvantageous, since this could result in an overload, which would greatly shorten the service life of the bearing.

## Advantages of the Invention

The electrical machine according to the present invention with the features of the main claim has the advantage that a large range of spring-force characteristics with a relatively flat increase in force is attainable. This means that the spring force and, as a result, the axial force load on the bearing may increase slightly

across the range of the intended compression of the spring element. As a result, the service life that can be expected of this bearing is markedly increased.

Advantageous further developments of the electrical machine according to the main claim are made possible by the measures listed in the subclaims.

Due to the fact that the spring element, in an outer region, bears against an outer ring of a rolling bearing and, in an inner region, against a hub projection, this results – in combination with an inner ring of the movable bearing being secured in a fixed manner on a rotor shaft – in a good application of load on this movable bearing in terms of the criteria for the usual design of rolling bearings.

According to a further subclaim, it is provided that the hub projection has a basically annular shape with a conical spring-support area that declines in the outward direction. As a result of these features, the spring element bears in a defined manner in the radially inward region, which also makes it possible to leave the effective leverage of the spring force unchanged or nearly unchanged. In addition, a simple tool geometry for manufacturing this surface against which the spring element bears is made possible, thereby also extending the service life of the tool. Due to the fact that the spring element configured as a disc spring essentially has the shape of a frustoconical shell, a high axial force can be achieved, so the rolling elements in the rolling bearing roll in their races under defined conditions. This results in a favorable service life for the rolling bearing.

According to a further exemplary embodiment, it is provided that a spacer is located in the force-transfer direction between the bearing and the spring element. This spacer, as does the hub projection provided with an outwardly declining conical spring-support surface, enables positioning in the hub without causing a change in the leverage of the axially-acting spring force. The bearing is therefore able to absorb axial vibrations relatively elastically without undergoing excessive changes in force.

It can also be provided that a spacer is located in the force-transfer direction between the spring element and the hub, that fulfills the same purposes.

The spacer can be, e.g., an originally separate ring fastened to a spring element, thereby resulting overall in an economical creation of a combination of spring element and spacer.

Drawing

Figure 1 shows a schematic illustration of a cross section through an electrical machine,

Figure 2a shows a cross section through a bearing for supporting a rotor using a spring element,

Figure 2b shows an enlarged depiction of a section of the hub projection,

Figures 3a and 3b show two views of the spring element in Figure 2a,

Figures 4a and 4b show, in a highly schematic view, the behavior of the spring element in Figure 3a and Figure 3b when loaded by axial vibrations of the rotor,

Figure 5a, 5b, and 5c show three different views of a second exemplary embodiment of a spring element,

Figure 6 shows a third exemplary embodiment of a spring element,

Figure 7 shows a cross section through the positioning sleeve in Figure 2a.

Description

Figure 1 shows, in a highly schematic view, a cross section through an electrical machine 10. Electrical machine 10 has, among other things, two housing parts 13 and 16, that accommodate, among other things, a stator 19. Housing parts 13 and 16 each have a hub 21 that serves to support shaft 25 of a rotor 27 via a bearing 23 and a bearing 24.

Left bearing 23 shown in Figure 1 is a “fixed bearing”, and bearing 24 shown in Figure 1 is a “movable bearing”. Movable bearing 24 and its configuration and position in hub 21 will be described in greater detail in conjunction with Figure 2.

Figure 2a shows, in a less schematic illustration, the arrangement of right bearing 24, movable bearing in hub 21, and bearing housing part 16. Housing part 16, often also referred to as an end plate, has hub 21 in its center, the hub extending axially in the shape of a cylindrical ring. A “hub projection” 30 adjoins hub 21, the hub projection extending radially inwardly. Hub projection 30 is located on the side of hub 21 facing away from rotor 27. Hub projection 30 has a basically annular shape and is formed by a few spokes 32, among other things, oriented in the radial direction. An annular spring-support surface 35 that is part of hub projection 30 adjoins spokes 32 further radially inwardly.

A “fitting ring 38” made of plastic is inserted in hollow-cylindrical hub 21. Fitting ring 38 serves to dampen vibration excitations between hub 21 and shaft 25. Shaft 25 extends into fitting ring 38, the shaft being held in fitting ring 38 by bearing 24. In the exemplary embodiment described, bearing 24 is designed as a rolling bearing, and specifically as a roller bearing in this case. The roller bearing is essentially composed of an inner ring 40, rolling elements 42 which are spherical in this case, and outer ring 44.

An axially acting spring element 47 is inserted between bearing 24 and hub projection 30. Spring element 47 is a disc spring that has an opening in its axial center that is provided for passage of shaft 25, which has a reduced inner diameter here. Disc spring 47 essentially has the shape of a frustoconical shell and, therefore, a radially inwardly directed inner region that bears against hub projection 30 and spring-support surface 35. Spring element 47 bears against outer ring 44 of bearing 24 with a radially outwardly directed outer region. Bearing 24, in turn, rests with its inner ring 40 on a shaft shoulder 50 of shaft 25.

To enable the most economical manufacture of the electrical machine possible, it has been provided that, during construction, a relatively great tolerance between spring-support surface 35 and left housing part 13 is permissible. It is also provided that the position of shaft shoulder 50 relative to spring-support surface 35 can be very different, so that, when bearing 24 bears against spring-support surface 35 via shaft shoulder 50 and spring element 47, very different, axially acting spring forces acted between spring-support surface 35 and bearing 24 when conventional, known spring elements are used. This is not desired. Rather, a range of spring-force characteristics with a relatively flat curve of force over time in the tolerance range is provided. For this reason, it is provided that the spring element configured as disc spring functions back and forth across a “flat” position of the spring. The flat position of spring element 47 is defined to mean that the outer region of spring element 47 has the same axial position as the inner region of spring element 47. This is synonymous with a shape of spring element 47 that now exists, which is nearly a plane.

An electrical machine 10 is therefore provided, designed in particular as a generator for motor vehicles, that has a rotatably supported rotor 27, whereby at least one bearing 24 serves to support rotor 27 in hub 21, and an axially acting spring force acts on bearing 24. Spring element 47 bears against hub 21 with spring force. Spring element 47 is a disc spring and is capable of functioning back and forth across the flat position of spring element 47.

Spring element 47 bears, in an outer region, against outer ring 44 of bearing 24 configured as a rolling bearing and, in an inner region, against a hub projection 30.

Hub projection 30 and spring-support surface 35 are shown in an enlarged depiction in Figure 2b. To better show the shape of spring-support surface 35, it was shown in this highly exaggerated illustration unequivocally as an outwardly declining, conical spring-support surface 35. The purpose of this highly conical spring-support surface 35 is to prevent an over-proportionally great increase in spring force between hub projection 30 and outer ring 44 during the axial vibrations of rotor 27. It is provided that, in the extreme position of spring element 47, i.e., in a particularly highly deflected position position of spring element 47 in the direction toward hub projection 30, spring element 47 does not come to rest against a radially inwardly lying edge 53 of spring-support surface 35. If spring element 47 were deflected to this extent, this would mean the spring force would no longer bear against the radially outwardly directed side of spring-support surface 35, but directly against edge 53, for example. As a result, the spring force between outer ring 44 and hub projection 30 would increase abruptly and bearing 24 or spring element 47 might therefore be overloaded.

For the same reason, it is provided, among other things, in the exemplary embodiment according to Figure 2a that, between outer ring 44 and the actual spring part of spring element 47, an annular spacer 56 is secured between the outer region of spring element 47 and outer ring 44. Without this spacer 56, spring element 47 could bear, in a position in which it is pressed through to hub projection 30, against an edge of the bore of inner ring 40. This would result in a reduction of the effective leverage of the axial force here as well and, as a result, the spring force would increase abruptly. To this end, spacer 56 is provided with conicity. Spacer 56 essentially has the shape of a frustoconical shell, the larger opening of which is directed toward spring element 47. The conicity is, e.g., 7° (the cone angle is therefore 14°). Spacer 56 therefore bears radially outwardly

with a substantially narrow, circular ring-shaped surface against spring element 47 and can also be connected with spring element 47 via this circular ring-shaped surface or be secured to it, e.g., using an integrally joining connection technique, such as welding, soldering or adhesive bonding. With this design, spacer 56 avoids a shortening of the effective leverage in the region of the transition from spring element 47 to spacer 56, which would occur with a flat spacer 56 and a simultaneously greatly deflected spring element 47. Otherwise, spring element 47 would bear against the radially inwardly located edge of spacer 56 directed toward spring element 47, resulting in an increased load on the spring element. A further reason for spacer 56 is that, due to spacer 56, it can be ruled out that spring element 47 would bear against inner ring 40 in the loaded and deflected state of spring element 47. Otherwise, there would be a risk that it would wear through. A spacer 56 is therefore located in the force-transfer direction between bearing 24 and spring element 47.

A view of spring element 47 from the left is shown in Figure 3a, based on Figure 2a. The view of spring element 47 shown in Figure 3b is a view from the right. The annular shape of spacer 56 and spring element 47 is clearly recognizable. Spring element 47 has an overall wave-shaped inner region. In this example, a total of three pegs 59 are formed on the outer edge of spacer 56, the function of which will be explained below in conjunction with Figure 7. The number of pegs 59 can also deviate from this; for instance, four or six pegs are also possible.

A schematic view of spring element 47 is shown in Figure 4a and Figure 4b. In both cases, the axial deflections of spring element 47 are shown in a greatly exaggerated manner to clearly illustrate the changes. Figure 4a illustrates the force and position relationships of a first operating state, in which a supporting force $F_1$ between spring-support surface 35 and spring element 47 acts on the radially inner side of spring element 47. An equal force $F_1$ acts between outer ring 44 and, in this case, spacer 56. Line $l_0$ was selected as the reference line, which, in this case, lies in the plane of osculation between spring element 47 and

spring-support surface 35. The axial height of spring element 47 between the side of spring element 47 facing hub projection 30 and the plane of osculation is $d_1$. The further extreme case in Figure 4b shows a spring element 47 deflected fully to the right past the flat position, whereby the side of spring element 47 facing hub projection 30 - the contact surface between outer ring 44 and spacer 56 - is now on the other side of flat position $1_0$, i.e., $d_2$. It is also clearly shown that force $F_2$, which is now acting between outer ring 44 and spring element 47, and between spring element 47 and spring-support surface 35 is nearly as great as force $F_1$ that was applied previously.

A further exemplary embodiment of a spring element 47 is shown in three different views in Figure 5a, Figure 5b and Figure 5c. Similar to Figure 3a, the illustration in Figure 5a is also a view from the left; accordingly, the view in Figure 5c is a view from the right, similar to the illustration in Figure 3b. As is the case with the exemplary embodiment of spring element 47 described above, this exemplary embodiment also has a radial inner region that has a generally wave-shaped configuration. The outer region of spring element 47 is slotted in close intervals and has alternating radial positioning elements 62 that serve to concentrically position spring element 47 in the fitting ring 38 and, therefore, to concentrically position the rotor axis. As an alternative, indirect concentric positioning can take place between spring element 47 and hub 21. Spacers 56 are located between the positioning elements, which bear against outer ring 44, as they do in the previous exemplary embodiments. Overall, these spacers 56 are curved and are U-bent at a nearly right angle.

Figure 6 shows a third exemplary embodiment of a spring element 47. It is provided that spring element 47 and spacer 56 for outer ring 44 are configured as a single component. In addition, but which can also be used individually, a further spacer 56 is located in the force-transfer direction between spring element 47 and hub 21, which is also integrally joined as a single component in this case.

Finally, a sectional view of fitting ring 38 is shown in Figure 7. Fitting ring 38 has insertion pegs 70 on one axial end, which are pressed between spokes 32 of housing part 16 and, in addition, as also shown in Figure 2a, are configured as snap-in elements and engage in grooves between spokes 32. Sections between pegs 70 also interact with spokes 32 such that spokes 32 acts as stops for fitting ring 38. Pegs 70 limit the play of fitting ring 38 in the axial direction toward the left, and the intermediate spaces between pegs 70 limit play toward the right, thereby resulting, overall, in a defined position of fitting ring 38 in hub 21. In addition, three grooves 73 are formed on the cylindrical inner circumference of fitting ring 38, into which pegs 59, mentioned above (refer also to Figures 3a and 3b) are inserted, thereby resulting in an overall unequivocal position of spring element 47, according to the exemplary embodiment in Figure 2a, 3a and 3b, in fitting ring 38. An incorrect position of spring element 47 in hub 21 is thereby ruled out. For the case in which an unequivocal match between spring element 47 and hub 21 is not required, pegs 59 and grooves 73 can also be distributed evenly around the circumference.